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Simon Webb

University of Wollongong

Zheng Li

University of Wollongong, zli@uow.edu.au

Christopher David Cook

University of Wollongong, chris_cook@uow.edu.au

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Computer Control of Machines with Friction for Precise Motion

Simon Webb, Zheng Li and Chris Cook
Department of Electrical and Computer Engineering
University of Wollongong
Northfields Avenue, Wollongong 2522 NSW
Australia

Abstract

In this paper we study the problem of precise control of machines with static and Coulomb friction. An experimental X-Y testbed built for testing various control methods is described and initial experimental results are presented. A simple pole placement controller is designed for computer control of the X-Y testbed based on a generalised sampled-data hold. Robustness and stability of the controller are analysed. Simulation suggests that this controller is able to achieve precise position control even when parameters of both the dynamics of the machine and the characteristics of friction are unknown.

Some of the important factors limiting the performance of modern servos include the effects of friction, especially about zero speed, cogging torques inherent in the construction of electric motors, flexing of the machine's frame and in the couplings, shafts and bearings in drive trains, backlash, and the paucity of position and velocity information available from encoders at very low speeds. The low speed problems are particularly severe when it is remembered that even though the part or tool in a multi-axis machine may be travelling relatively rapidly, one or more axes may at times pass through zero speed. For example, each axis of a two-axis machine tracing out a circular path of fixed radius will pass through zero speed twice per revolution. Any non-linear movement or 'lost motion' caused by stiction or backlash will detract from the accuracy of the circle, and reduce process quality.

1 Introduction

Machine tool builders are under ever increasing pressure to improve the position and velocity tracking ability, accuracy, and process quality available from their machines. For example, accuracies down to microns or better are becoming the rule rather than the exception, and machine tool purchasers are increasingly expecting their machines to be able to produce improved quality or mirror finishes on even the most complex parts.

Consequently the ability of each of the servos controlling a machine's axes to accurately follow velocity trajectories and precise position commands is central to the machine's overall performance. Also the ability to incorporate additional information from sensors into servo control loops potentially provides improved process quality. One example of this is the incorporation of automatic torque control into machine tools used for grinding.

Friction plays an important role in high precision motion control because of its high nonlinearity at very low velocities. Tracking errors are immediate results of the so called stick-slip limit cycling caused by friction in a closed-loop system where two machine parts keep sticking and sliding alternately with respect to each other. Precise control of machines with friction is receiving considerable attention in the area of control systems engineering. For a recent survey on modelling and control of friction and related topics see [1].

Theoretical and experimental investigations on computer control of machine tools under the influence of friction are sometimes carried out on an X-Y testbed. For example, accelerated evolutionary programming has been applied for friction identification and control of a seven parameter friction model, and experimental results show the elimination of stick-slip in positioning control for very low speed tracking [2]. In [3], repetitive control was applied for low speed friction compensation for

machine tool feed drives, where static friction is dealt with as a periodic disturbance while the X-Y testbed is repeatedly following a circular trajectory. The contouring error is asymptotically reduced through learning passes, wherein the controller output is iteratively adjusted to compensate for the tracking errors from previous cycles.

Impulsive control has been developed for precise position control, where static friction plays an important role [4]. Its idea is to apply a series of small impacts when the system is at rest so that each will cause a small displacement. Applying a series of small impacts with correctly adjusted energy the system can be moved bit by bit and finally stop at the desired position with high precision. A major advantage of this method is that it does not require detailed information about friction. The energy required by the next impact can be determined using a learning mechanism based on the observation of the displacements caused by the past and current impacts. It has been shown that the effects of friction can be greatly reduced. In particular, stiction is overcome because the amplitude of the impact can be chosen to be higher than static friction such that a displacement is guaranteed.

This paper reports on work being conducted on the problems of precise motion control under the influence of friction as part of a research project within the Cooperative Research Centre in Intelligent Manufacturing Systems and Technologies. The industrial partner in this work is ANCA, one of the world's leading manufacturers of precision grinding machines. This paper describes the construction of an instrumented testbed to allow the testing of precise motion control algorithms, reviews some of the different ways of potentially dealing with friction via feedback control and presents some preliminary results for computer controller design and observations based on practical tests carried out on the testbed.

2 Testbed construction

This section describes the testbed's mechanical hardware, and its computer control capability and the instrumentation provided. The testbed is intended to provide a construction which mimics as accurately as possible the layout, drive trains, bearing design and servo systems of a modern machine tool. This allows the very low speed characteristics of a practical machine to be measured, accurately modelled, and then the ability of various innovative control schemes to improve

performance to be quantitatively compared with a benchmark commercial 'state-of-the-art' controller.

Since a machine with two axes will test many vital characteristics of a multi axis machine, the testbed shown in Fig. 1 has been constructed. The actuators are rare earth permanent magnet Baldor AC servos (BSM80A) which provide, in this system, a maximum short-term torque of 8.3 N-m. These drive precision double nut 5mm pitch ball screws with preload via toothed belts and pulleys as shown. The carriage is mounted on ball bearing type linear bearings. Maximum motor speeds are 1500 rpm and positional accuracy to 1 micron is provided. The construction method provides compactness but the pulleys contribute most of the inertial load on the motors, even when the carriage is fully loaded.

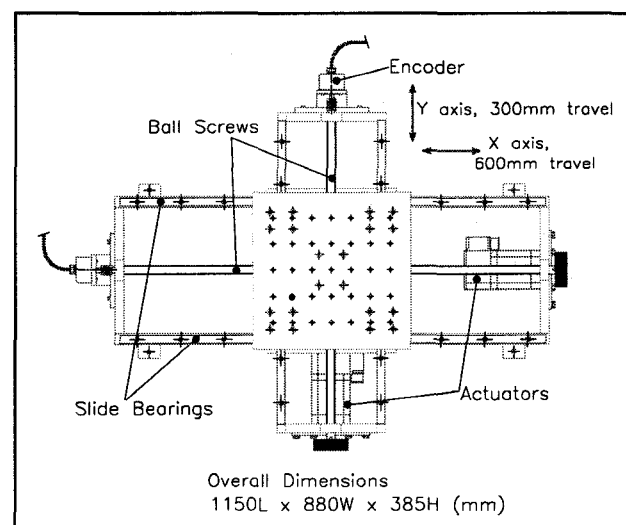


Fig. 1 Plan view of testbed

A block diagram of the testbed is given in Fig. 2. The actuators are driven by Baldor servo modules (DBSC 105) with sufficiently fast current loops (1kHz bandwidth) that, when connected to produce torque directly proportional to an analogue input signal, they can for most practical control system purposes be treated as simple proportional amplifiers. Their input signal is controlled from a digital signal processor in which position, velocity and force or other control schemes can be implemented by the experimenter. The Baldor modules can also be reconfigured, under software control from the Pentium personal computer, to provide velocity control together with pole placement compensator design. In this configuration the DBSC receives velocity feedback from the resolvers built into the servomotors.

The diagram illustrates the architecture of the computer hardware interface module. A dashed box encloses the core components: DSP μP , DSP Bus/Memory, PC μP , DSP Digital I/O, DSP Analogue I/O, and PC14 Digital I/O. The DSP μP is connected to the DSP Bus/Memory via a bidirectional arrow. The PC μP is connected to the DSP Bus/Memory via an ISA Bus. The DSP Bus/Memory is connected to the DSP Digital I/O via a DSPLink and to the DSP Analogue I/O via a bidirectional arrow. The PC μP is connected to the PC14 Digital I/O via an ISA Bus. The DSP Digital I/O is connected to the Encoder Interface Module via Polling Signals and Data. The DSP Analogue I/O is connected to the Servo Module DESC 105 via an Analogue Command. The PC14 Digital I/O is connected to the Signalling Interface Module via Drive Enable and Drive Status Signals. The Encoder Interface Module is connected to the Servo Module DESC 105 via Encoder Evaluation. The Servo Module DESC 105 is connected to the Signalling Interface Module via a bidirectional arrow. The Servo Module DESC 105 is connected to the X-Y Table Module via Limit Switch Information. The X-Y Table Module is connected back to the Servo Module DESC 105 via a bidirectional arrow.

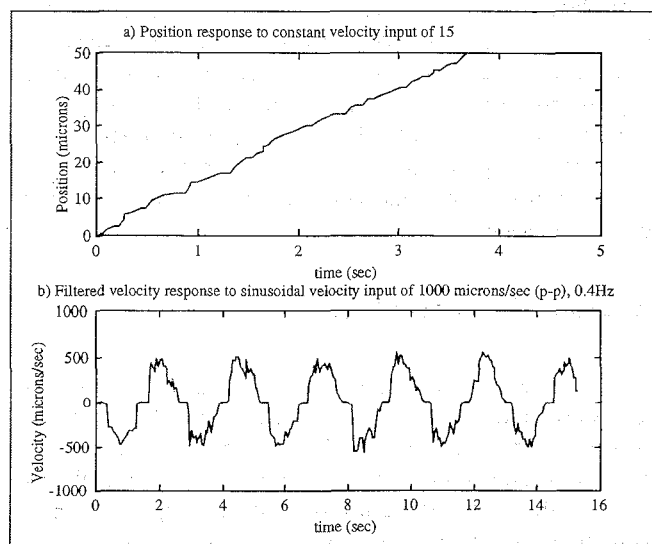
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graph TD
    DSPUP[DSP μP] <--> DSPBus[DSP Bus/Memory]
    PCUP[PC μP] <-- ISA Bus --> DSPBus
    DSPBus -- DSPLink --> DSPDIO[DSP Digital I/O]
    DSPBus <--> DSPAIO[DSP Analogue I/O]
    PCUP <-- ISA Bus --> PC14DIO[PC14 Digital I/O]
    DSPDIO -- "Polling Signals and Data" --> EIM[Encoder Interface Module]
    DSPAIO -- "Analogue Command" --> SM[Servo Module DESC 105]
    PC14DIO -- "Drive Enable and Drive Status Signals" --> SIM[Signalling Interface Module]
    EIM -- "Encoder Evaluation" --> SM
    SM <--> SIM
    SM -- "Limit Switch Information" --> XYM[X-Y Table Module]
    XYM <--> SM
  
```

High level planning, control, monitoring, trajectory generation and data processing is provided by the Pentium PC. Considerable effort has been made to ensure that this provides user friendly commands, prompts and displays to facilitate the testbed's operation, data capture and processing. All real-time control is carried out at very high speed by the Loughborough Sound Images DSP, which can provide sample rates at each axis of up to 200kHz.

troubleshooting and setting up of the motors and amplifiers can be conveniently performed using Baldor's proprietary software running under DOS on the PC. However, most experimental programming is performed in C and Visual C++ is used for setting up control screens, calculating trajectories and high level planning and precalculation of trajectories. Real-time control programs can be developed in C on the Pentium PC and then loaded into the DSP. Data, such as speed and velocity transients, is collected by the PC and can either be analysed immediately or transferred into files that can be processed by MATLAB. Thus the development and operating environments are very convenient, and provide for quick development, analysis and comparison of control schemes, and also allow the full force of MATLAB to be accessed for modelling, design and analysis purposes.

Fig. 3 illustrates position and velocity response at low speed of the X-axis using experimental data obtained from the testbed. Stick and slip phenomena can be clearly seen in both the position and velocity response. In position response the stiction is manifested as horizontal lines and in the velocity response it is indicated by the zero speed periods at the cross over velocity.



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3 Pole placement controller

The dynamic equation of the plant to be controlled including friction is described by:

$$\ddot{x} = \begin{cases} \frac{1}{M}(f - f_c) & \text{if } \dot{x} \neq 0 \\ 0 & \text{if } \dot{x} = 0 \text{ and } |f| \leq f_s \\ \frac{1}{M}[f - \text{sign}(f)f_s] & \text{if } \dot{x} = 0 \text{ and } |f| > f_s \end{cases} \quad (1)$$

where M is the equivalent mass, x is the position of the X-Y testbed on the X axis, f , f_s and f_c are the input force, static friction and Coulomb friction with f_s and f_c positive and $f_s > f_c$, $\text{sign}(\cdot)$ is a sign function defined as: $\text{sign}(y)=1$ if $y>0$, $\text{sign}(y)=-1$ if $y<0$ and $\text{sign}(y)=0$ if $y=0$.

We combine the pole placement method [5 Astrom and Wittenmark] and a generalised sampled-data hold function for position control of the X-Y testbed. The generalised sampled-data hold proposed is shown in Fig. 4, where f_1 , f_2 and f_3 are amplitudes of pulses applied to the X-Y testbed as driving forces with pulse widths T_1 , T_2-T_1 and $T-T_2$. T is the sampling period and $T_1 \leq T_2 \leq T$. The idea of impulsive control is extended in this paper using this sampled-data hold for use at very low velocities such that the machine comes to a complete stop at the end of each sampling period allowing the simplification of friction and system dynamics in discrete time. In this sampled-data hold we use the driving force as controller output rather than a voltage. The reason is that the dynamics of the electrical drive are much faster than that of the mechanical part of the X-Y testbed and can, therefore, be ignored when the sampling period is long enough.

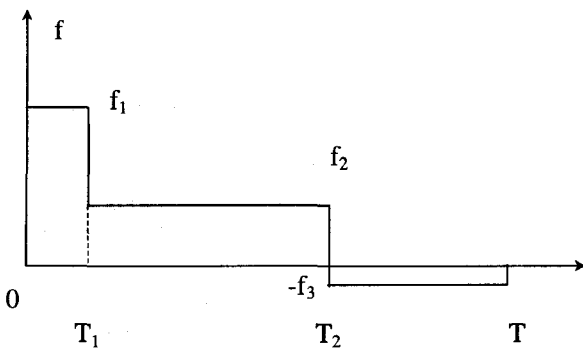


Fig. 4 A generalised sampled-data hold

The amplitude of the first pulse satisfies $f_1 > f_s$ such that static friction is overcome and a displacement is guaranteed. It is assumed that $f_2 \geq f_c$ which enables a further displacement in the desired direction. The relocation of some energy from the first pulse into the second allows a smoother motion. The amplitude of the third pulse is negative in order to make the machine stop completely before the end of the sampling period. Its amplitude satisfies $f_3 < f_c$ such that the possibility of a displacement opposite the desired direction is avoided. While the amplitudes are fixed the instants T_1 and T_2 are variable and related by $T_2-T_1=\alpha T_1$ where α is a constant.

Following the method in [4], we find the displacement caused by the sampled-data hold during a sampling period and the discrete-time transfer operator of the plant to be controlled by integrating equation (1) with input as shown in Fig. 4. The displacement for one impulse is given by

$$X = CT_1^2 \quad (2)$$

where

$$C = \frac{1}{2M} \{ (1 + 2\alpha)(\Delta f_{c1}) + \alpha^2(\Delta f_{c2}) + \frac{[\Delta f_{c1} + \alpha \Delta f_{c2}]^2}{f_c + f_3} \}$$

with $\Delta f_{c1} = f_1 - f_c$ and $\Delta f_{c2} = f_2 - f_c$.

Noting that the machine comes to a complete stop at the end of each sampling time and that $X = x(k+1) - x(k)$ we write down its dynamics in control system notation:

$$x(k) = \frac{K}{q-1} u(k) \quad (4)$$

where $K=|C|$, q is the one-step-ahead operator and the input is:

$$u(k) = T_1^2 \text{sign}(C) = T_1^2 \text{sign}(f_1) \quad (5)$$

The system "input" is defined to be $u(k)$, a function of the impulse width of f_1 with the same sign (Because the width of f_2 is proportional to that of f_1 the input determines the entire shape of the sampled-data hold). This way of defining $u(k)$ linearises the system and so greatly simplifies the design and analysis. Given the desired closed-loop transfer operator

$$x(k) = \frac{T(q)}{q^2 + cq + d} r(k) \quad (6)$$

where $r(k)$ is the reference input to the closed-loop system, the pole placement controller has the form

$$(q-1)u(k)=T(q)r(k)-(aq-b)x(k) \quad (7)$$

where $q-1$ on the left-hand side of the equation introduces integral action to reduce steady state error, $a=(c+2)/K$ and $b=(1-d)/K$. The closed-loop system is shown in Fig.5, where SDH stands for the generalised sampled-data hold.

The root locus of the closed-loop system is given by:

$$(aK) \frac{q-b/a}{(q-1)^2} = -1 \quad (8)$$

where the double poles of the open-loop system are fixed at $+1$ and the zero is at b/a . The root loci for $b/a = 0.9, 0.3, 0$ and -0.3 are shown in Fig. 6. All the loci on the circles are complex numbers. The smallest circle is part of the root locus for $b/a=0.9$ and the largest circle is for $b/a=-0.3$. The unit circle is part of the root locus for $b/a=0$. Clearly, when the open-loop zero is a negative real number the circular part of the root locus will be outside the unit circle and the root locus on the real axis will have at least one root outside the unit circle for any K . Consequently, the closed-loop is unstable for any K . When $b/a > 1$ the root locus will be on the real axis over the interval $(-\infty, b/a)$ and there will always be a closed-loop pole in $(1, b/a)$ causing an unstable closed-loop system. For $0 < b/a < 1$ the circular part of the root locus is entirely inside the unit circle and the root locus on the real axis spans the entire negative real axis and a part of the positive real axis inside the unit disc. The closed-loop system is stable when the absolute value of the real parts of the closed-loop poles are less than unity. Substituting $q=-1$ into equation (8) we have the condition for closed-loop stability,

$$0 < b/a < 1$$

$$K < \frac{4}{a+b} \quad (9)$$

For the above two conditions the choice of a and b is in our hands and K is determined by the inertia of the system, static friction and Coulomb friction. Noting (9) we need only to know that K is bounded (always true in practice) to make the closed-loop stable. Once the upper bound is known it is always possible to choose appropriate a and b to satisfy condition (9).

In simulation we choose $a=2$, $b=1$ and $T(q)=q$. The closed-loop system has the form

$$C(q) = \frac{Kq}{(q-1)^2 + K(2q-1)}$$

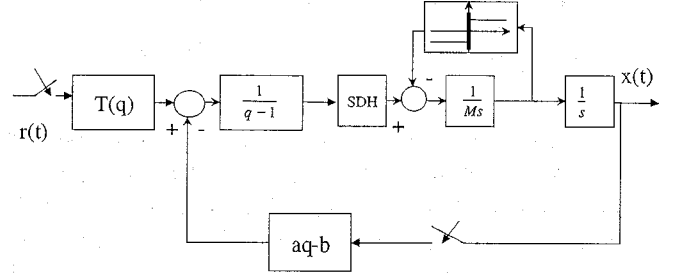


Fig. 5 Closed-loop control system

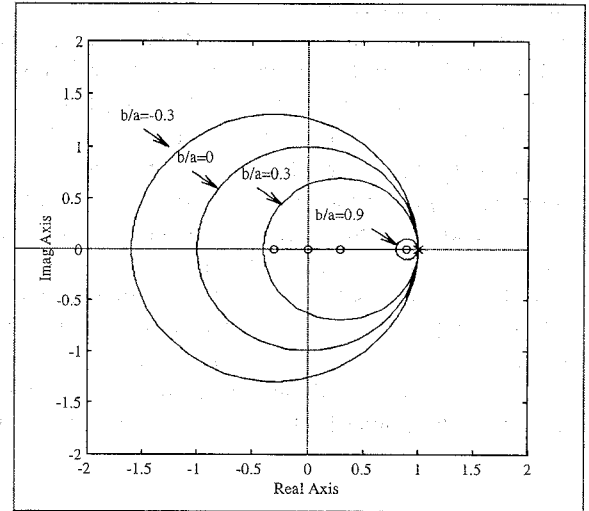


Fig. 6 Root locus for the pole placement controller

When $q=1$ we have $C(1)=1$ which is independent of K implying zero steady state error without requiring any knowledge about the parameters of the system and the friction. Simulation results are illustrated in Fig. 7, where $K=0.75$, the reference signal $r(k)$ is the desired position consisting of a ramp followed by a constant represented using a solid line, the actual output is presented using circles. It clearly shows that the position of the testbed follows the reference signal on the sampling instants in the steady state.

4 Conclusion

The design and construction of a testbed for precise motion control together with its associated computer control and instrumentation has been presented. This testbed has mechanical construction and bearing arrangements similar to a modern machine tool. Initial experimental results are given to illustrate how nonlinear friction effects can provide stick-slip operation in position and velocity motions. Advanced computer control is to be used to overcome this limit cycling and a computer controller has been designed for position control of the X-Y testbed based on pole placement strategy and a generalised sampled-data hold function. The potential of this controller has been demonstrated using simulation, where stick-slip motion in position regulation is avoided, even though negligible information about both the dynamics of the X-Y testbed and the friction is required. As an extension to impulsive control the need for parameter estimation [4] for adaptive control is eliminated here using pole placement control with appropriately chosen parameters. Further research is in progress as the authors continue to develop and test computer controllers for the testbed.

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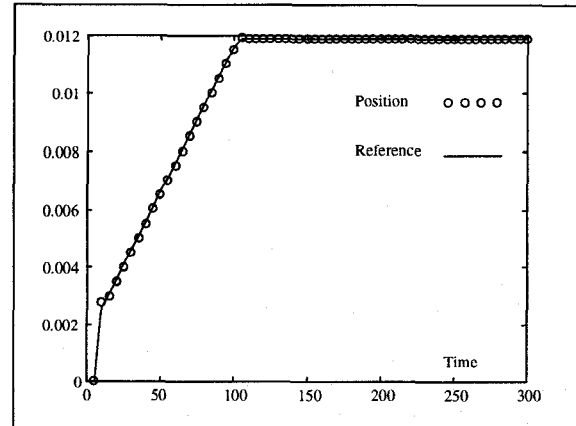


Fig. 7 Simulation results